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APR 85 DAAK70-82-K-0222

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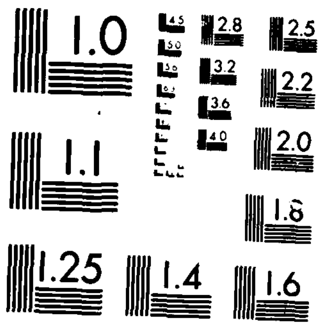

MICROCOPY RESOLUTION TEST CHART
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Source: Ceres Corporation, Waltham, MA 02154
Title: Linear Resonance Cooler
Contract No. DAAK-70-82-C-0222
Date: April 1985
Controlling Office: AMC ERADCOM Night Vision and Electro-Optics Laboratory, ATTN: DELNV-SI, Fort Belvoir, VA 22060

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FOREWORD

This report documents the efforts of designing, fabricating and testing three Linear Resonance Split Cryocoolers.

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This report was prepared by Graham J. Higham, Dr. Peter J. Kerney and John Vigilante.

1. INTRODUCTION

CTI-CRYOGENICS has currently completed a program of work which was funded by the U.S. Army Night Vision and Electro-Optics Laboratory under Contract DAAK 70-82-C-0222 to design, fabricate and test three Linear Resonance Split Cryocoolers. The contract requires the incorporation of linear drive technology into a cryocooler design meeting the ~~B2~~ Specification ~~HD-1045~~ (V)/UA for the US Army 1/4 Watt Common Module Cooler as closely as possible.

Present day cryogenic coolers employ the conventional rotary motor, crankshaft and reciprocating piston designs in the compressor and either direct or pneumatically driven displacers in the expander or cold end. Contamination from greases, motor windings and other contaminant sources inherent in the design of these cryocoolers present serious problems that have plagued the producibility and reliability of these devices. Acoustic noise and self induced vibrations produced by the aforementioned coolers are also of concern with many system applications.

The CTI-CRYOGENICS Resonant Cryocooler ~~(Figure 1)~~ employs a concept using magnetic forces to linearly move dual opposed pistons in the compressor thus eliminating the use of rotary motors, crankshaft, greases and bearings. While not a requirement of the contract, the expander also employs a linear motor to control displacer stroking. This allows for electronic end stopping to mitigate microphonics problems. In addition, the ability to vary the waveform of the displacer motion and its phasing with respect to the compressor pressure wave form in order to optimize performance offers the potential for extending the operational life of the cooler.

Contained in this report are the results of the program which comprises:

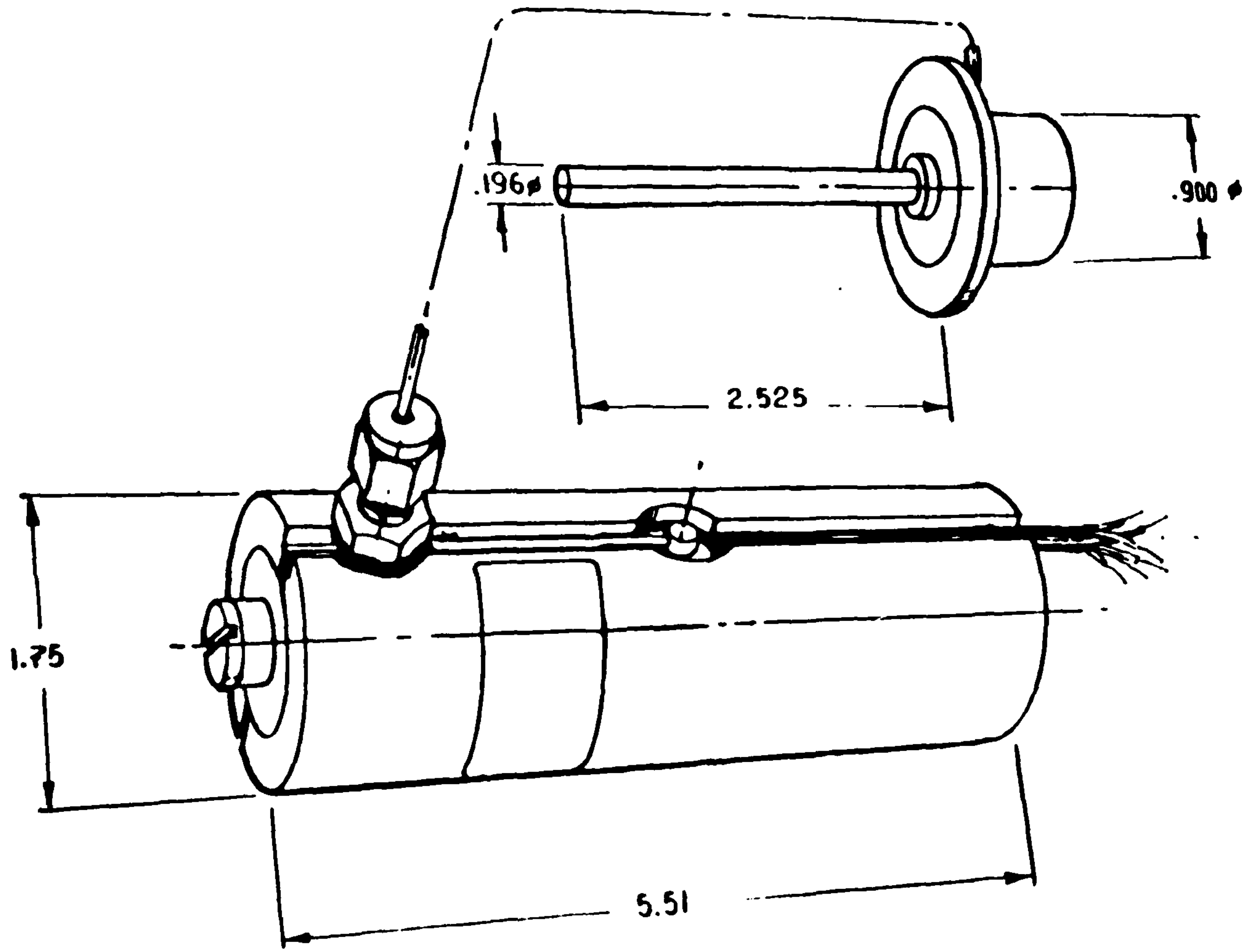


FIGURE 1. LINEAR RESONANCE CRYOCOOLER

- the testing of an experimental linear motor driven expander using a standard production 1/4W split Stirling Common Module compressor.

- the detailed design of the Linear Resonance Cryocooler.

- Cryocooler performance testing.

2. DETAILED DESIGN OF LINEAR RESONANCE CRYOCOOLER

2.1 General

CTI-CRYOGENICS elected to preserve the thermodynamic design (bore, stroke, fill pressure, frequency etc.) of the existing 1/4 watt Common Module Cryocooler. This minimized the program risk and allowed for a clearer assessment of the contributions of the features described in the foregoing.

For a resonant system, as the frequency increases the mass of the rotor and piston have to be much smaller and are governed by the relationship:

$$C = mw^2$$

where m = the total moving mass
w = operating frequency, and
c = overall elasticity constant due to the gas compressibility.

Although there are weight and overall volume benefits in operating at higher frequency our choice to operate at 25 Hz, the same frequency as the current 1/4 watt common module cryocooler, proved to be very beneficial in allowing CTI-CRYOGENICS to adopt a proven production thermodynamic design to a linear drive concept.

2.2 Expander Design

CTI-CRYOGENICS has long recognized the potential of employing a linear drive motor to assist regenerator displacement and phasing for improved performance and extended life. At the time of this contract award, CTI-CRYOGENICS had already been developing linear drive technology and demonstrating the feasibility of the displacer assist principle. A standard production Common Module expander was adapted for coupling to a linear drive motor and the unit was tested using a standard production compressor.

Subsequent work performed under the contract with the U.S. Army NV&EOL included testing of an experimental linear drive expander. This expander incorporates a linear drive motor in a design which is essentially the same as the Common Module expander and meets the envelope requirement of the HD-1045(V)/UA Specification. In addition, it was designed to employ either lip or clearance seals interchangeably for comparative evaluations. Tests of this expander were conducted using a standard production Common Module compressor. The results of these tests, which are discussed in the following section (3.2), were used to optimize the design of the expander for the linear resonance cooler to be supplied under this contract.

To guarantee performance the salient thermodynamics of the expander are identical to the current 1/4 Watt Common Module Cooler produced by CTI-CRYOGENICS. Figure 2 shows a cross sectional view of the expander. The design meets the envelope specification of the HD-1045(V)/UA envelope by physically locating the linear motor rotor within the expander's pneumatic volume. A Hall sensor, mounted between the motor stator windings provides positional information for the electronic servo system. Incorporated within the design, and satisfying the contractual requirements of the purchase

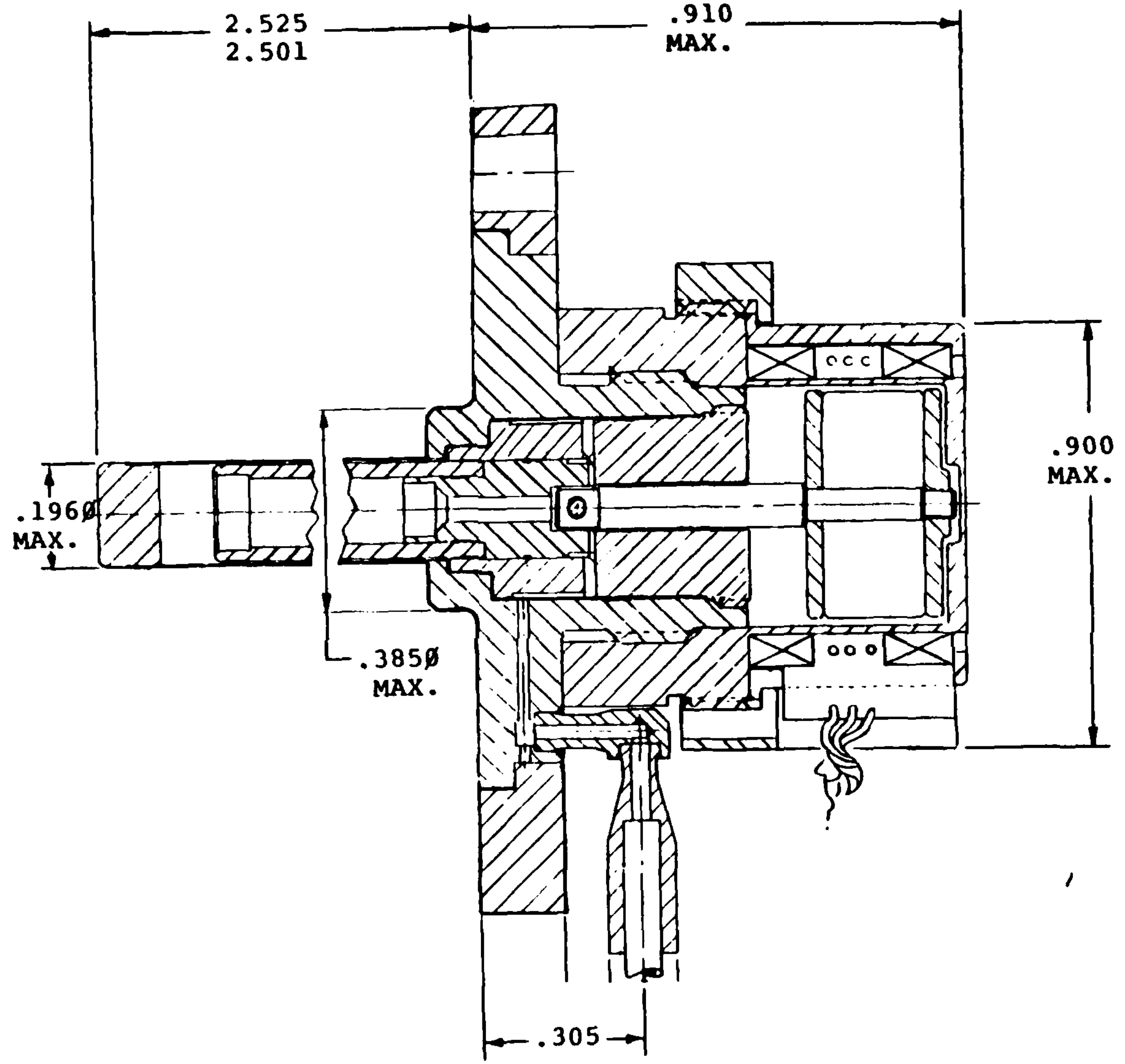


FIGURE 2. LINEAR DRIVE EXPANDER

description, are clearance seals for both the drive piston and displacer seal.

2.3 Compressor Design

The compressor consists of two opposed piston/motor rotor assemblies reciprocating within a common compressor cylinder. Each compressor piston/rotor mass and size has been optimized to take advantage of the resonance aspects of the gas spring. Also the design of the motor and the size and stroke of the compressor pistons are designed to meet the required compressor performance.

A drawing of the linear resonance cryocooler compressor is shown in Figure 3. Each motor has its own Hall effect sensor to sense piston position thus enabling satisfactory servo control. These sensors are mounted between the motor coil assemblies and together with the coils are external to the working helium gas environment. Hence, no electrical feed thrus are necessary with this design. Helium integrity is accomplished with the use of indium seals and metal 'C' seals. Clearance seals are used as the dynamic piston seals. Clearance gap and wear of these seals have been addressed by choosing compatible materials with similar expansion coefficients and minimum wear rates.

As shown in Figure 3, the charging port extends outside the envelope specified for the compressor per HD-1045(V)/UA. This exception could have been circumvented, but to do so would have required a more complex mechanical design. It was felt that the additional risk inherent in that approach was not justified in the design of a prototype cooler. All other dimensions, including the location of the compressor transfer line fitting, are in compliance with the specification.

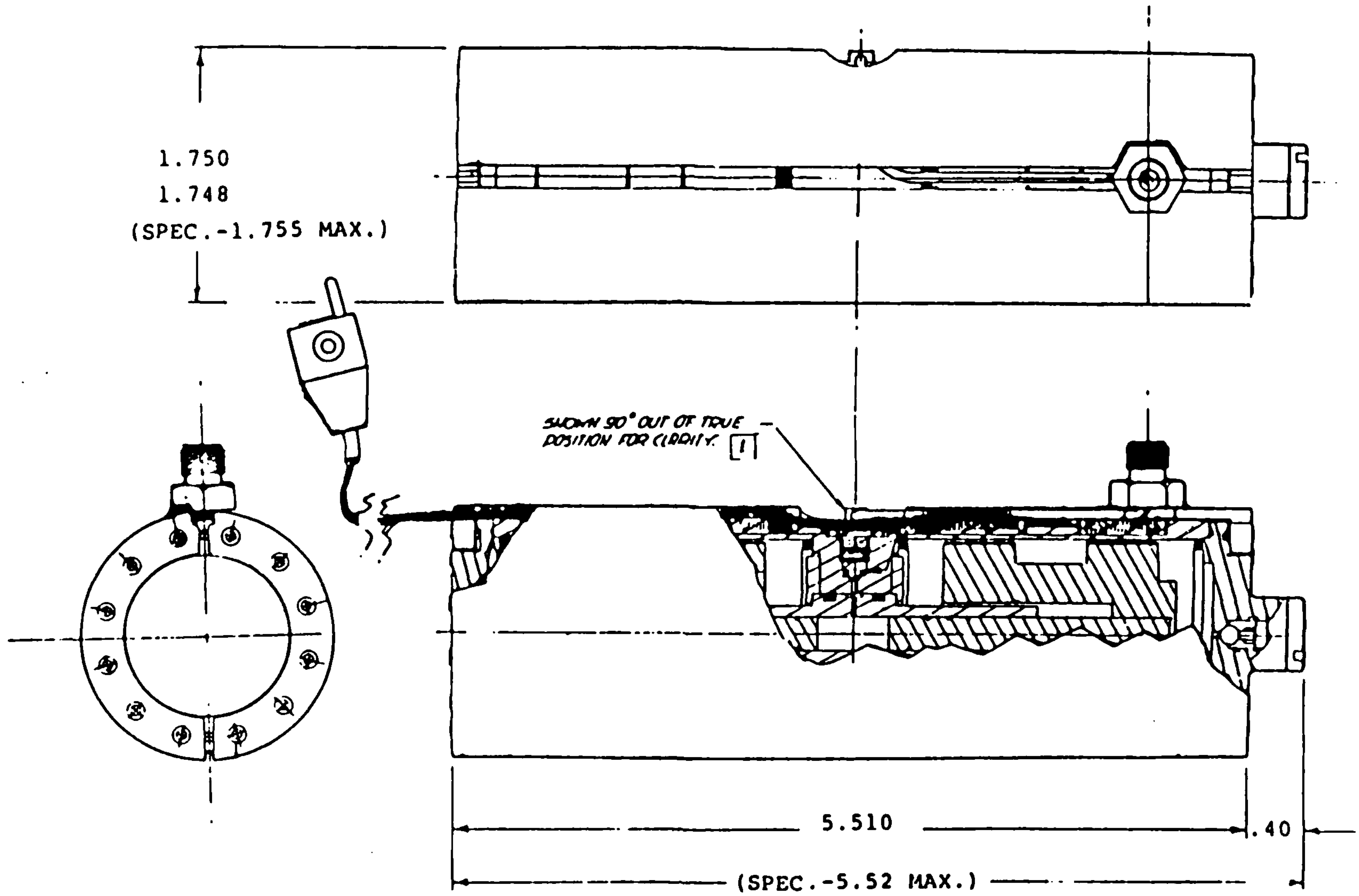


FIGURE 3. LINEAR RESONANCE COMPRESSOR

2.4 Size and Weight

Apart from the charging port the compressor and expander has been designed so that their sizes comply with the development specification for the HD-1045(V)/UA cooler. Figures 2 and Figure 3 show how the cryocooler design meets the envelope requirement.

Table 1 compares the weight and volume breakdown of the 1/4 Watt Common Module Cooler and the linear drive cryocooler. The weight measurement of the linear drive cooler shows that it meets the weight requirement of 2.5 lbs (measured value 2.44 lbs)

2.5 Electronic Controls

The control system for the compressor must maintain precise phase relationships between each independent compressor power piston and a pacing waveform. In addition to the phase requirement the control system must enhance the resonant characteristics of the compressor arrangement as well as maximizing stroke in a limited displacement area. The compressor control technique evolved from a preliminary linear design to a more advanced nonlinear approach.

The expander control system implements a variety of force contours that achieve phase or dwell control and end stopping of the displacer. A Hall effect sensor provides position information to the control system and positional changes are effected through the use of a linear motor attached to the displacer.

2.6 Compressor Electronic Control

In the preliminary compressor control system Hall effect devices with limited compensation were used to sense piston position. The Hall device signal contains a stator excitation component as well as the position component. The

	1/4 WATT COMMON MODULE CRYOCOOLER		LINEAR DRIVE CRYOCOOLER	
	WEIGHT (LB)	VOLUME (IN^3)	WEIGHT (LB)	VOLUME (IN^3)
EXPANDER	0.0971	0.43	0.155 lbs	0.47
COMPRESSOR	1.991	12.94	2.266 lbs	13.23
TRANSFER LINE	0.021	---	0.021	---
TOTALS	2.11	13.37	2.442	13.70

TABLE 1. COOLER WEIGHT AND VOLUME COMPARISON

stator excitation component is eliminated and the remaining signal is compared with the pacing waveform to create an error signal. The error and the integral of the error are used to control the motor drive level. Analog techniques are used to take the error signal and control power to the motor.

The final compressor control scheme (Figure 4) uses active Hall effect subsystems (LOHET) These LOHET'S have Hall effect devices in addition to local amplification and temperature compensation. To obtain consistent armature position information throughout stroke the average of two LOHET'S is used. The armature magnets are segmented and produce angular non-uniformities. The motion of the armature takes a swirling form as the piston and armature stroke. The sensory confusion between angular displacement and linear displacement is eliminated with this averaging technique. The average output signal is then processed by a lowpass filter which eliminates the stator excitation component. This clean piston position signal is compared with a pacing waveform to generate an error signal. The error, error-rate, and integral of the error are used to modulate power to the motor of interest.

A pulse width modulation (PWM) scheme is used to vary compressor motor power. Varying the compressor motor power with a PWM subsystem will provide maximum motor drive efficiency. The PWM subsystem has an internal free running oscillator. This high frequency oscillator creates a pulse train time base. The servo error signal then modulates this pulse train time base. The time average of the modulated pulse train is propotional to the servo error signal. The output of the PWM subsystem controls a set of transistors switches. This set of transistor switches interrupts current through the stator windings in the compressor motor.

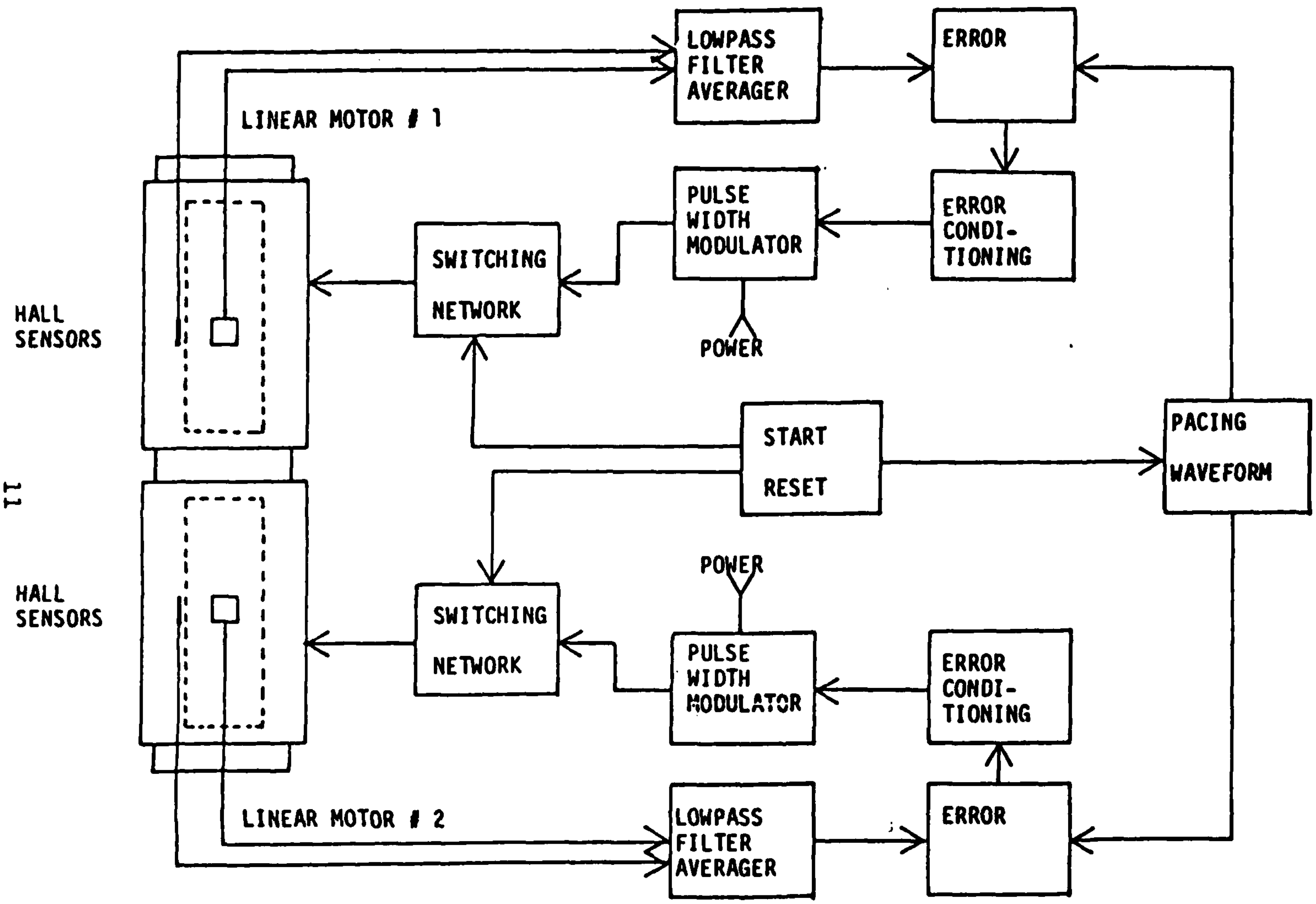


FIGURE 4. COMPRESSOR ELECTRONIC CONTROL DIAGRAM

This situation lends itself to excellent signal discrimination between the piston motion frequency and the stator excitation frequency. These two frequencies are orders of magnitude removed, and therefore easily filtered. The motor direction is controlled by a switching network that senses the sign of the error signal. The control scheme has provisions for an organized startup or reset condition.

2.7 Expander Electronic Control

The expander control scheme (Figure 5) requires a pressure wave reference signal from the Compressor. The piston position within the compressor provides an accurate pressure wave facsimile. In a situation when displacer phase offset is required, the control scheme accepts a phase shift set point. The expected displacer position as a function of the pressure wave is summed with the phase shift set point. This summation output is a phase shifted reference wave and is fed forward to the input of a phase locking system. The actual displacer position is obtained with a Hall effect sensor in the displacer linear motor. The displacer position satisfies the feedback information needed by the phase locking system. The phase locking system itself maintains drive levels to the displacer linear motor to achieve any required repositioning.

The Hall effect sensor in the displacer linear motor is externally compensated and provides position, velocity and acceleration information. End stopping requires knowing the displacer acceleration near the physical end stop locations. The linear motor is pulsed with a reversing signal that is proportional to the surplus acceleration at that position.

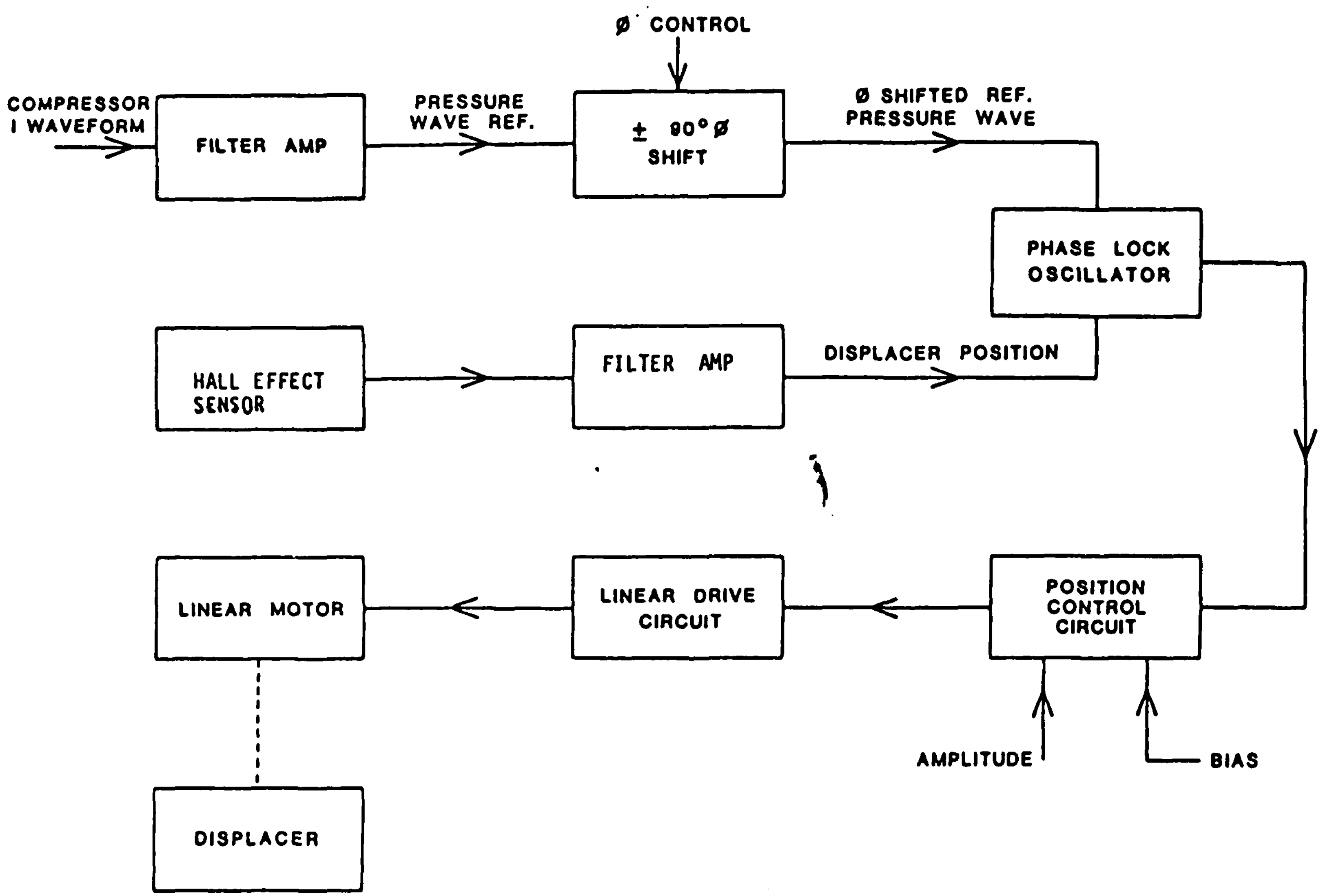


FIGURE 5. EXPANDER ELECTRONIC CONTROL DIAGRAM

3. EXPERIMENTAL TESTING

3.1 General

An experimental program was pursued to:

- validate our analytical modeling techniques by quantifying and confirming compressor and expander performance as independent entities and as a system.

- evaluate the performance of the cryocooler at different ambient temperatures and to determine the reliability of the cryocooler.

3.2 Expander Performance

3.2.1 Output Refrigeration Performance Characteristics

Output refrigeration performance characteristics were established to formulate a baseline assessment of the expander's performance with lip seals and clearance seals for a regenerator comprised of nickel balls. Further comparison of a stainless steel mesh regenerator was also made. The effect of the linear drive motor on the cooler performance for these combinations of sealing methods and regenerator materials was evaluated. As an example, Table 2 illustrates the beneficial action of the linear drive on cooler performance for a nickel ball regenerator.

On concluding this experimental expander phase of the program, implementation of displacer clearance seals into the three coolers to be supplied under this contract were made. This type of seal was a contractual requirement of the program.

	OUTPUT REFRIGERATION PERFORMANCE AS A % OF 1/4W COMMON MODULE BASELINE PERFORMANCE *
LIP SEAL (WITH MOTOR)	142%
CLEARANCE SEAL (NO MOTOR)	121%
CLEARANCE SEAL (WITH MOTOR)	152%

COLD TIP TEMPERATURE	80K
REGENERATOR MATERIAL	NICKEL BALL
AMBIENT TEMPERATURE	25°C

* 1/4W COMMON MODULE
BASELINE PERFORMANCE
0.25W AT 60K

TABLE 2. LINEAR DRIVE EXPANDER PERFORMANCE EVALUATION COMPARISON

3.2.2 Expander Electronic End Stopping

Expander electronic end stopping was achieved by using the expander motor as a brake at the moment prior to the displacer hitting its end stop. With this method the kinetic energy of the displacer is absorbed by the motor. During the dwell time at either end of its travel the displacer rests on its end stop. This negates the need for power to be consumed by the motor during this time.

Figures 6 and 7 are actual data illustrating the end stop capability. An accelerometer was mounted on the warm end of the expander to monitor the vibration of the displacer as it rapped at either end of its travel. It can be seen from this data that displacer end rapping can be virtually eliminated by correctly phasing suitable electrical power signals to the motor.

The size of the braking pulses are a measure of the dynamic force excess acting on the displacer. This excess will change with cold tip temperature or applied heat load; Table 3 illustrates this variation. It can be seen from this table that the expander motor adequately provided for the electronic end stop function with only a relatively small amount of the electrical input power. No degradation of cryocooler performance was detected when the expander motor was providing the electronic end stopping.

The comparative size of the braking pulses at each end of the displacer stroke are a measure of the correct sizing of the drive piston for the particular displacer/regenerator configuration being tested. Figure 7 illustrates this point, where it can be seen that a greater motor power/force is needed at one displacer end stop than the other. Performance of this

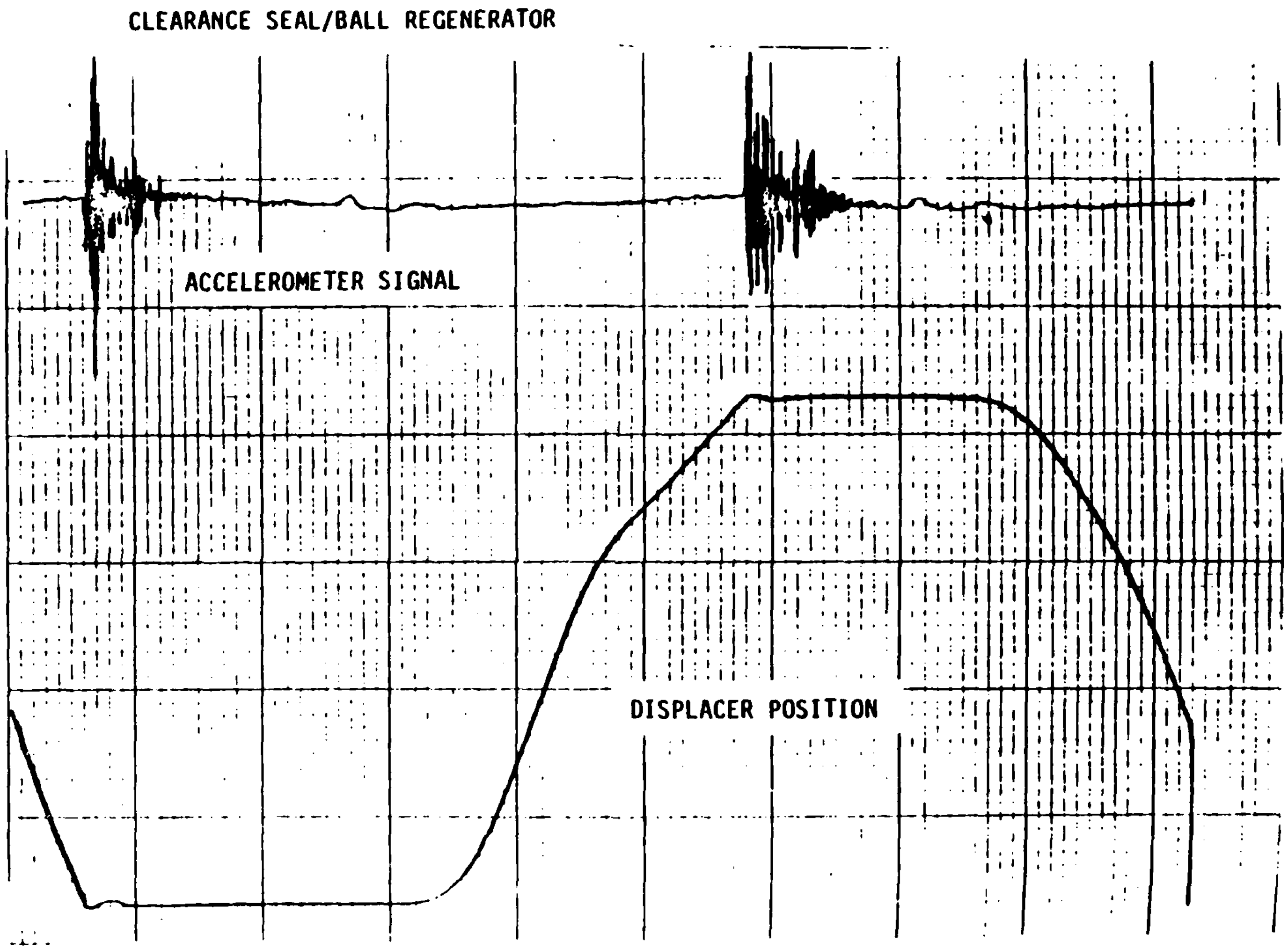


FIGURE 6. DISPLACER RAPPING WITH NO MOTOR CONTROL

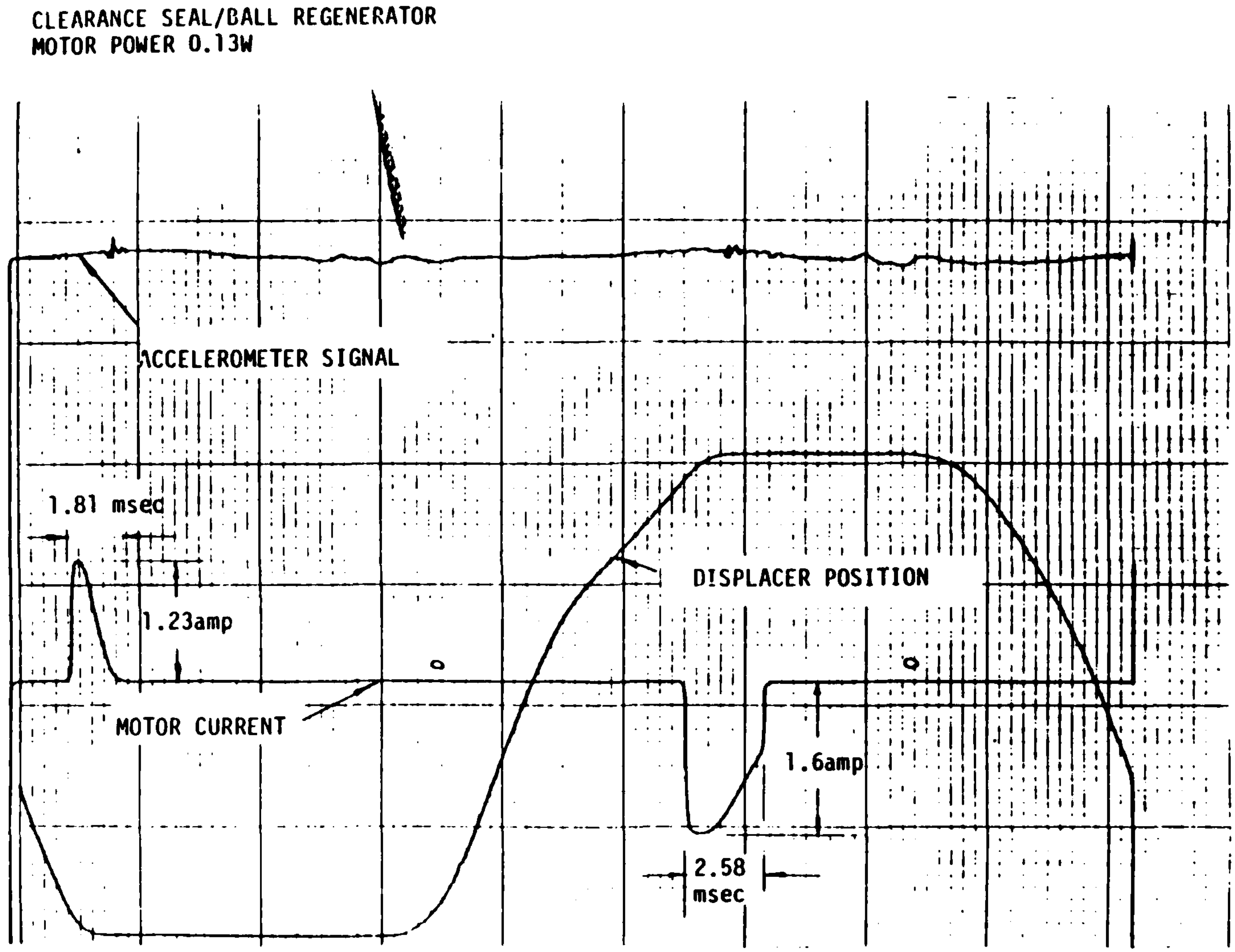


FIGURE 7. ELIMINATION OF DISPLACER RAPPING WITH MOTOR CONTROL

EXPANDER LINEAR MOTOR POWER REQUIREMENT TO EFFECT END STOP CONTROL (WATTS)	APPLIED HEAT LOAD (WATTS)
0.01	0
0.12	0.25
0.22	0.44
0.31	0.65

TABLE 3 ELECTRONIC END STOP: LINEAR MOTOR POWER REQUIREMENT
AS A FUNCTION OF APPLIED HEAT LOAD
(AMBIENT TEMPERATURE 25°C)

cooler was not degraded due to this problem but could ultimately be affected during the life of the cooler where displacer friction and regenerator pressure drop variation could possibly occur. These combined conditions could result in displacer short stroking which would cause a significant degradation in cryocooler performance. As a result of this particular observation, and as this test work was performed on a prototype expander, suitable corrective action was made to the sizing of the drive pistons of the three cryocoolers supplied under this contract. This size adjustment amounted to only a small change in piston size; a change that resulted in a negligible phase change between the displacer and compressor piston motions.

3.2.3 Phase Shift Control

To date, with the current electronic control logic and the present expander drive motor design, a limited amount of phase shift control has been accomplished. Currently, it is possible to achieve a displacer phase shift of ± 13 degrees for a maximum input power of 2.5 watts. Figure 8 shows displacer phase shift as a function of power to the expander linear drive motor. This phase shift control was also performed with electronic end stops. A ball regenerator was used for the above tests. Tests were also performed with a mesh regenerator, with similar results.

3.2.4 Varying Dwell Time

Dwell time is defined as the time per cycle that the displacer remains at either end of its stroke. This action changes the shape of the displacer motion. Preliminary tests with the experimental linear motor expander has shown the capability of controlling this dwell time with the linear motor to be possible.

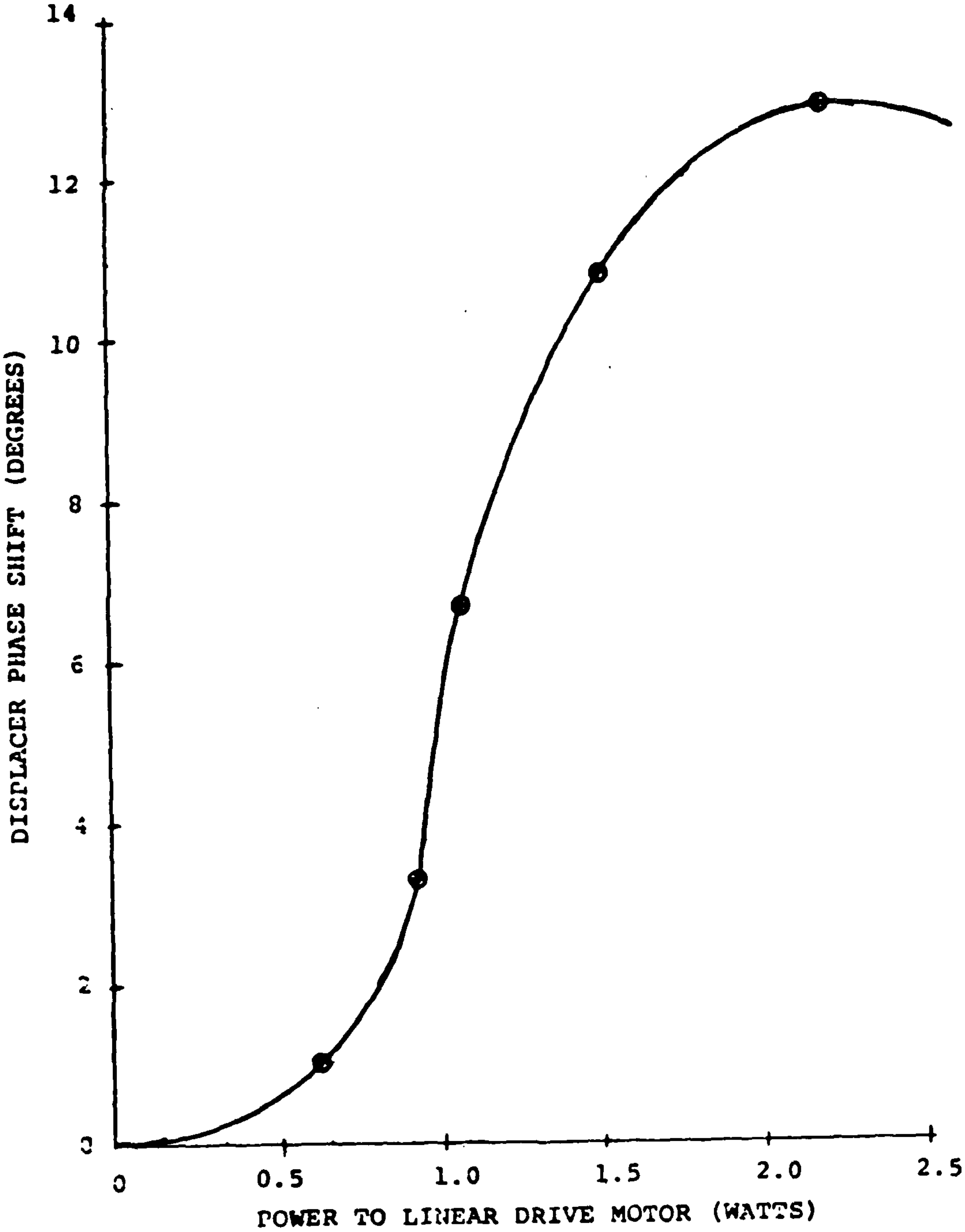


FIGURE 8. DISPLACER PHASE SHIFT AS A FUNCTION OF POWER TO EXPANDER LINEAR DRIVE MOTOR

At the end of the experimental expander test phase it was decided to simplify the operating function of the linear drive expander. In order to simplify the expander electronic design it was decided, at NV&EOL's request, to retain only the electronic end stop capability for buffering and to discontinue, for the present, the ability of the expander motor to tune performance thru phasing and dwell time control.

Figure 2 shows a cross sectional view of the expander design. The linear motor is physically located in the pneumatic volume thus enabling the envelope specification of the HD-1045(V)/UA to be met. A Hall effect sensor, mounted between the motor stator windings provides positional information for the electronic servo system.

3.3 Compressor

Prior to the final assembly of each compressor, calibration of each Hall effect positional transducer was performed. The calibration basically involved moving each rotor assembly with precision micrometers and recording the output from each sensor. The expected output should have been linear with respect to the linear position of each rotor. However, nonlinear signals were experienced on two of the compressors when their respective rotors were at their extremes of stroke. As a result it was not possible to achieve the necessary compressor pressure wave performance with each of these units and poor cryocooler performance was experienced. This particular problem will be explained in greater detail in section 3.6.

Each compressor was subjected to the same output pressure wave test that is currently being conducted on production, CTI-CRYOGENICS manufactured, 1/4 watt cryocooler compressors.

This test involved monitoring the pressure wave from a pressure transducer mounted on a known void volume. The compressor that was used in the subsequent detailed system testing passed this test successfully. The other two compressors due to the deficiency already described above were unable to pass this test in their present form. This period of testing also enabled rigorous evaluation of the electronics prior to the mating of each compressor to its respective coldfinger. Helium leak testing of these units showed that our expectation of meeting this aspect of the contract requirement could be achieved.

3.4 System Parametric Testing

In order to perform a comparative evaluation of the linear resonance cryocooler, a development 1/4 Watt Common Module (CM5) cryocooler with clearance seals was acquired and the following combinations tested at room ambient temperatures were conducted:

- linear drive compressor and expander
- CM5 compressor and expander (with clearance seals)
- linear drive compressor and CM5 expander
- CM5 compressor and linear drive expander.

The results are plotted in Figure 9. As can be seen from this figure, the linear drive cryocooler performance compared very favorably with that of the CM5 cryocooler, indicating sound thermodynamic performance. Furthermore, Figure 10 shows the performance of the linear resonance cryocooler when compared with production CTI-CRYOGENICS manufactured CM5 cryocoolers.

3.5 System Performance and Life Testing

Apart from the load curve performance data (see Figures 9 and 10), continuous monitoring of compressor output pressure as well as compressor piston and displacer position were

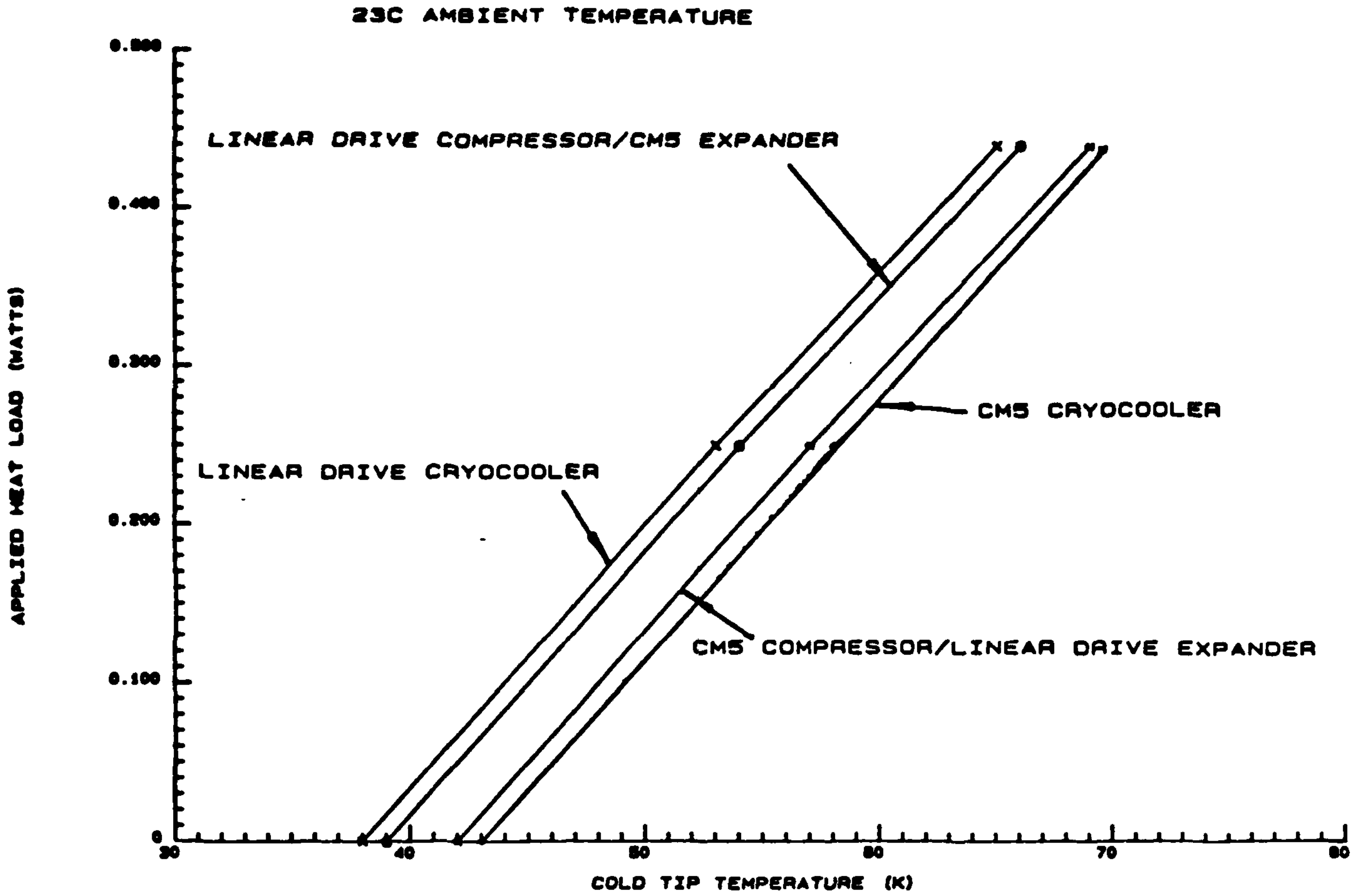


FIGURE 9. LINEAR DRIVE/CM5 PERFORMANCE COMPARISON EVALUATION

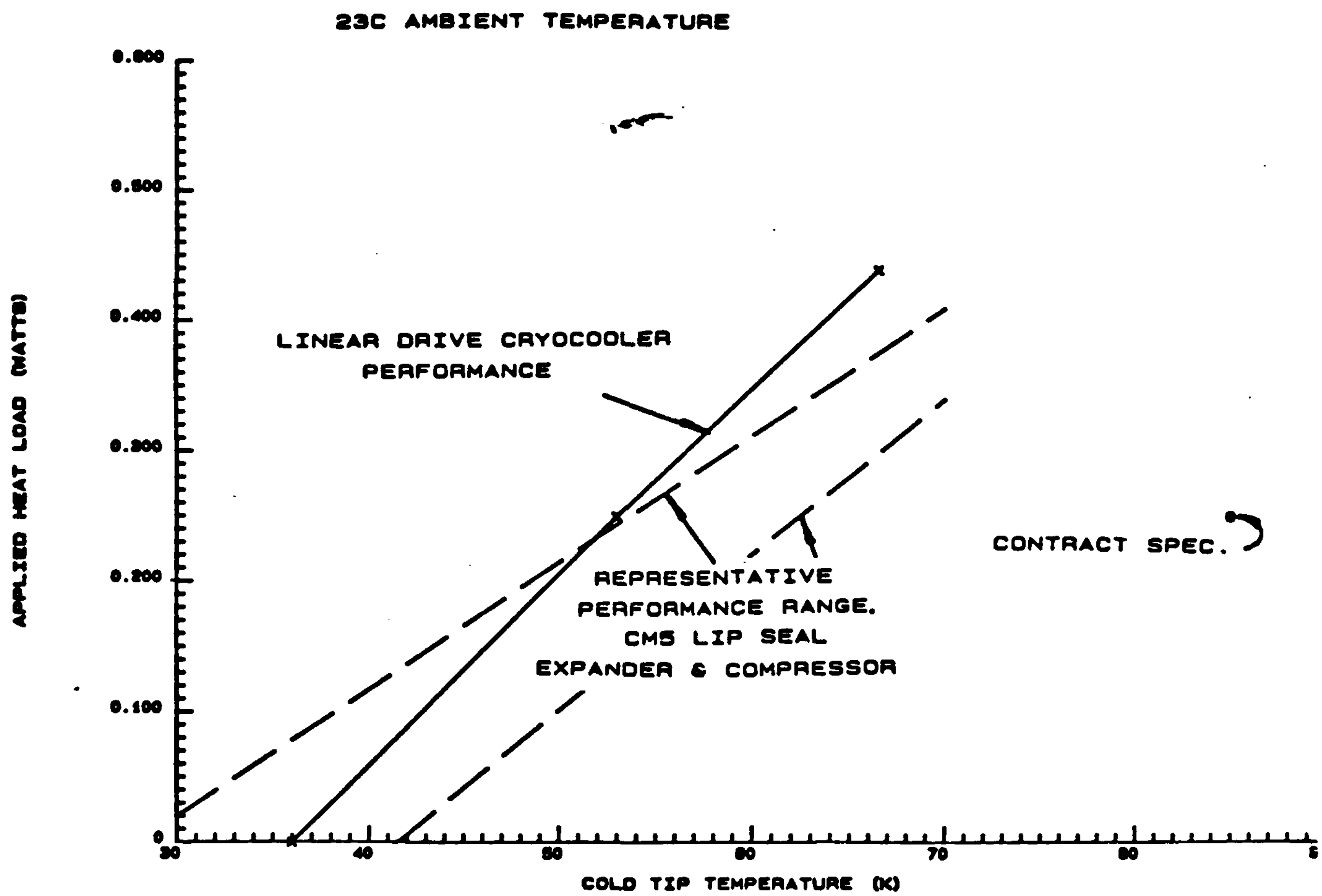


FIGURE 10. COMPARISON OF PERFORMANCE WITH CM5 CRYOCOOLER

recorded. This data together with the electronic excitation waveform to the drive motors were stored in an oscilloscope for later display onto an x-y plotter. Figure (11) shows an example of the type of data displayed on the oscilloscope. This figure shows the various harmonic positional and pressure waveforms that were recorded, together with the compressor P-V diagram and a diagram of the motor output force as a function of rotor axial position. Alongside the oscilloscope record are performance parameters that were determined from this information data. This collective data enabled detailed analyses of the various adjustments made to the electronic circuitry and also permitted optimum adjustments to be made. Figure 12 shows a less cluttered data record without the various positional and pressure waveforms and takes the form:

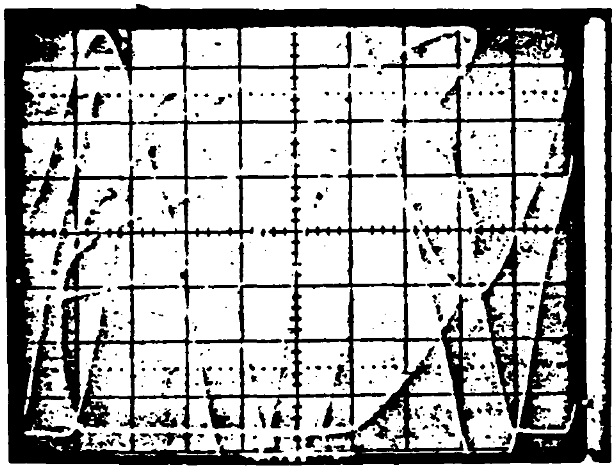
- Compressor pressure as a function of compressor piston position. This particular figure could be readily modified to become a P-V diagram, the integration of which would be the compressor output power. Numerous integrations of data recorded this way were made to validate our thermodynamic model.

- Expander warm end pressure as a function of displacer position. Once again this figure can be interpreted as a P-V diagram, the area of which is a reflection of the cryocooler output power or refrigeration. As can be seen in Figure 12, the relative position of the two P-V diagrams gives a measure of the phase angle between the compressor pistons and the displacer. The phase angle of 98 measured here is close to the theoretical optimum for a Stirling cycle cryocooler.

COMPRESSOR MOTOR FORCE
VERSUS ROTOR AXIAL POSITION

COMPRESSOR P-V DIAGRAM

COMPRESSOR MOTOR
VOLTAGE VERSUS
TIME



COMPRESSOR PRESSURE
VERSUS TIME

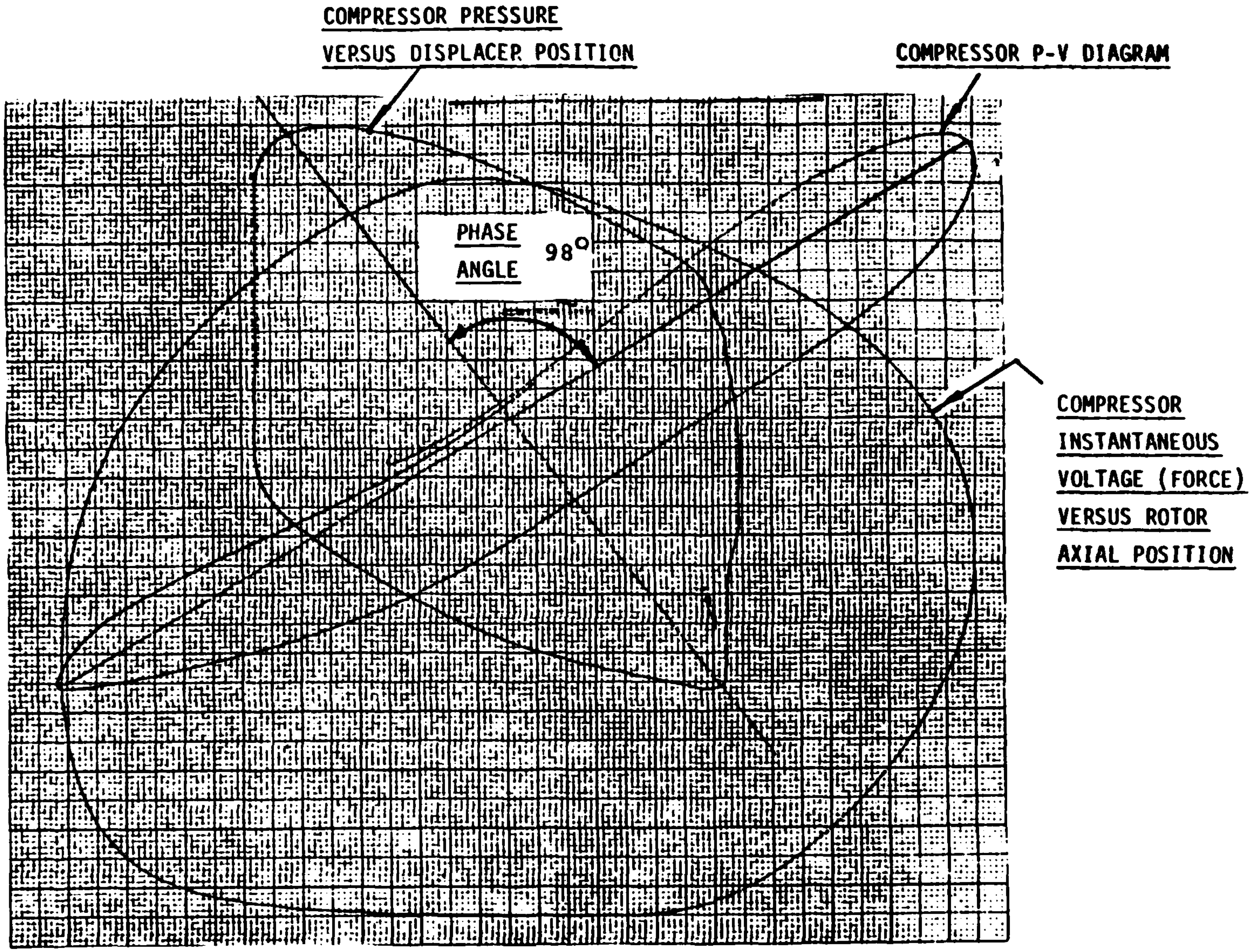
CRYOCOOLER PERFORMANCE

COLD TIP TEMPERATURE	70°K
HEAT LOAD	0.25 WATTS
AMBIENT TEMPERATURE	77°F
INPUT POWER	28 WATTS

DATA DETERMINED FROM ABOVE FIGURE

(1) GAS SPRING CONSTANT
(2) MOTOR PEAK FORCE
(3) INPUT POWER
(4) COMPRESSOR OUTPUT POWER
(5) PHASE ANGLE - MOTOR FORCE TO ROTOR POSITION

FIGURE 11. OSCILLOSCOPE TRACE: CRYOCOOLER PERFORMANCE



COLD TIP TEMPERATURE 70°K, HEAT LOAD 0.25 WATTS
AMBIENT TEMPERATURE 77°F, INPUT POWER 28 WATTS

FIGURE 12. CRYOCOOLER PERFORMANCE

- Compressor instantaneous voltage as a function of compressor piston position. Input power to each motor can be determined by calculating the area of this diagram. The shape of this diagram is close to being elliptical and occurs when the peak motor forces are at their minimum. At this condition the motor I^2R losses are at their minimum hence the condition of minimum input power and resonance.

  A computer was used to determine the area of this input power diagram and hence enabled areal time assessment of the adjustments made to the drive to the linear motors. Apart from providing optimization ability this information also confirmed the electronic input power measurement technique.

Difficulty in effecting satisfactory expander linear motor control resulted in a delay in system environmental temperature testing. These difficulties will be expounded in more detail in section 3.6. As the expander problems could not be corrected quickly, it was decided to acquire a CM5 cryocooler expander with clearance seals and run this with the linear drive cryocooler compressor. Two environmental temperature tests were conducted with the cryocooler in this configuration.

The first environmental test was terminated after 30 hours due to instrumentation problems. The second test was run for 3 1/2 days (84 hours) with the ambient temperatures cycling in compliance with the purchase specification. Testing was terminated due to poor high ambient temperature performance. Table 4 tabulates the details of this test and compares the performance with production CM5 cryocoolers. Very consistent performance

LINEAR DRIVE CRYOCOOLER COMPRESSOR WITH CM5 EXPANDER (CLEARANCE SEALS)
HEAT LOAD 0.25 WATTS

	LINEAR DRIVE		CM5	
AMBIENT TEMPERATURE °C	COLD TIP TEMPERATURE K	INPUT POWER (WATTS)	COLD TIP TEMPERATURE K	INPUT POWER (WATTS)
23	68	26	65	28
52			69	26
72	86	23		
-20			58	26
-40	56	24		

3 1/2 DAY (84 HOUR) TEST - LESS THAN 1°K TEMPERATURE VARIATION AT EACH AMBIENT TEMPERATURE CONDITION.
TEST TERMINATED DUE TO POOR HIGH AMBIENT TEMPERATURE PERFORMANCE.

TABLE 4. CRYOCOOLER RELIABILITY AND PERFORMANCE EVALUATION

was maintained during this test with no detectable cryocooler cold tip temperature drift variation at each of the ambient temperatures. Also the input power to the cryocooler was maintained well below the required specification of less than 30 watts. The deficiency at high ambient temperature has since been considered to be due to compressor piston short stroking as a result of erroneous positional information from the Hall effect positional transducer at this temperature. This being the case would also explain the reduction of cryocooler input power at this temperature.

3.6 Design Modifications and Problems

During the experimental testing phase, the following problems were encountered that resulted in the need to implement certain design changes.

- <u>Windage Losses</u>
  The original compressor design had too small a radial clearance between each rotor and its housing that resulted in considerable windage losses. This condition also affected the dynamics of the rotor drive which together resulted in input power in excess of 50 watts. This problem was addressed by increasing the radial gap which resulted in a significant reduction of overall input power.

- <u>Expander Motor</u>
  Performance of the three expander motors, fabricated for this contract, did not perform to the same expectations that was experienced with the prototype expander design. Thermodynamically the expanders performance was acceptable as demonstrated when performance data was acquired when the expanders were assembled without the linear drive motors. On installing the motors into these

expanders, displacer short stroking was experienced. The probable cause for this problem was side loading, due to radial misalignment accentuated by the inability to ensure concentricity of the polarizing flux path of the motors permanent magnet with respect to its flux return path. This problem could be corrected with certain mechanical design modifications.

- <u>Hall Effect Positional Transducer</u>
  Difficulty in ensuring a linear output from the Hall effect positional transducer over the full range of the compressor piston stroke was a serious problem. System testing of only one compressor was possible due to this nonlinearity.

  Hall sensor temperature compensation was also a problem with the one compressor that was system tested. After extensive testing of this compressor it became apparent that it was not possible to guarantee satisfactory sensor performance over the full operational temperature range of the cryocooler. Testing was conducted with two different types of sensors, one was temperature compensated the others not. This particular problem could be addressed by designing the necessary temperature compensation into the circuit control board.

4. CONCLUSION

Table 5 gives a summary of the Linear Drive Cryocooler Performance comparison characteristics compared with the contract design goals.

TECHNICAL SPECIFICATION	REQUIREMENT	MEASURED OR ACTUAL CHARACTERISTICS
Cooler Type	Split Stirling	Split Stirling
Motor Type	Linear Drive	Linear Drive For Both Expander and Compressor
Electronics	Not Specified	Breadboard, externally mounted for 3 prototype units Hybridized integrated system for production
Cooling Capacity	>0.25 Watt @ 85K in 68° ambient >0.25 Watt @ 85K between -40°F and +150°F	Compliant
Cooldown Time	7.5 min or less to 100K in 68°F ambient 10 min or less to 80K in 68°F ambient Thermal mass 120 Joules	Compliant
Maximum Power Consumption	30 Watts	Less than 28 Watts

TABLE 5. SUMMARY OF COOLER PERFORMANCE

TECHNICAL SPECIFICATION	REQUIREMENT	MEASURED OR ACTUAL CHARACTERISTICS
Emitted Acoustic Noise	To be determined	To be determined
Helium Leak Rate	2.7×10^{-7} scc/sec	2.7×10^{-7} scc/sec for 3 prototype units Compliant with HD-1045(V)/UA for production
Vibration Output	To be determined	To be determined
Weight	2.5 pounds	2.44 pounds
Size	Desirable to meet configuration of HD-1045(V)/UA	Compliant Except for Compressor Fill Port
Life	MTBF 2500 Hours	Expect to meet life specification when design deficiencies are corrected.

TABLE 5. SUMMARY OF COOLER PERFORMANCE (CONINUED)

Deficiency of the cryocooler to meet the required performance at high ambient temperatures precluded the cryocoolers ability to demonstrate its life and reliability potential. However, the limited test program has demonstrated the application of linear motor drive technology to a Stirling cycle cryocooler design. Also, the compressor demonstrated the potential benifits of minimal vibration and input power of a dual opposed compressor operating at resonance conditions. The problems outines do not appear to be insurmountable and the work so far accomplished has provided a firm foundation to continue in this area of technology.

Virtually all the major deficiencies of current rotary drive compressors have been addressed by the development of this cryocooler. It is expected that once the problems so far encountered are resolved, a cryocooler exhibiting high reliability and long life will result.

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